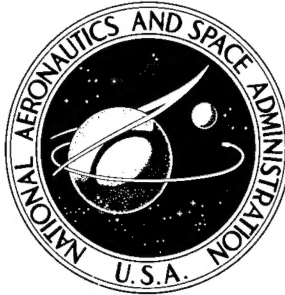


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COOLING REQUIREMENTS OF
BALL BEARINGS LUBRICATED
BY GLASS-FIBER-FILLED
POLYTETRAFLUOROETHYLENE
RETAINERS IN COLD HYDROGEN GAS

by Harold H. Coe, David E. Brewe, and Herbert W. Scibbe

Lewis Research Center

Cleveland, Ohio

DEPARTMENT OF DEFENSE
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SUMMARY

An experimental investigation was conducted to determine the cooling requirements of 40-millimeter-bore ball bearings operating in 60° R (33 K) hydrogen gas. A 15-weight-percent glass-fiber-filled polytetrafluoroethylene (PTFE) retainer material was used for lubrication. The bearings were run at speeds of 20 000, 30 000, and 35 000 rpm with thrust loads of 200, 300, and 400 pounds force (890, 1335, and 1780 N). The measured hydrogen-gas coolant flow rates ranged from 0.008 to 0.045 pound mass per second (0.0036 to 0.0204 kg/sec) through the bearings.

An empirical equation was developed to represent the minimum flow rate of coolant gas required by the bearings for thermally stable operation for the range of operating conditions investigated. The minimum flow results with this retainer material were very similar to results obtained previously with a 30-weight-percent bronze-filled PTFE retainer material.

Postrun inspection of the bearings showed that the retainer wear rate was very low; the average rate was about 0.03-percent weight loss per million sliding feet (0.3×10^6 m) at the retainer locating surface for all but one of the nine bearings tested. This bearing showed heavy wear in two ball pockets, and the retainer wear rate was about 0.3 percent per million sliding feet (0.3×10^6 m). Some inner-race ball-track wear was evident on most of the bearings that had been run for more than 7 hours.

INTRODUCTION

Turbopumps are used in rocket engines to deliver liquid fuel at high pressures to the thrust chamber. To save weight and improve reliability, it is desirable to operate

the turbopump bearings directly in the propellant, and thus eliminate an external lubrication system (refs. 1 and 2).

Ball bearings operated in hydrogen, however, normally require a self-lubricating material for the retainer to provide sufficient lubrication (refs. 2 and 3). Of the retainer materials that have been evaluated for cryogenic applications, two of the most successful have been 30-weight-percent bronze-filled polytetrafluoroethylene (PTFE) and 15-weight-percent glass-fiber-filled PTFE (refs. 3 and 4). In reference 4, bearings using these two materials for retainers ran for 10 hours in 60° R (33 K) hydrogen gas. The bearings were 40-millimeter bore, operating at a shaft speed of 20 000 rpm and a thrust load of 200 pounds force (890 N). Retainer wear was light, and inner-race wear was not discernible.

Cooling of the bearing is provided by the cryogenic hydrogen itself, either liquid (refs. 2, 5, and 6) or gas (refs. 4, 7, and 8). To utilize the hydrogen efficiently, the minimum flow rate required to achieve thermally stable operation of the bearing must be known.

In reference 7, cooling studies were made to determine experimentally the relation between the bearing outer-race temperature and the hydrogen coolant-gas flow rate. Tests were conducted with 40-millimeter-bore ball bearings with a 30-weight-percent bronze-filled PTFE retainer, operating in hydrogen gas at 60° R (33 K). Minimum coolant flow rates were also determined over a range of test conditions.

The objectives of the present investigation were the following:

(1) Determine experimentally the relation of the bearing outer-race temperature with hydrogen-gas coolant flow rate for bearings with a 15-weight-percent glass-filled PTFE retainer.

(2) Determine the minimum coolant flow rate required with the 15-weight-percent glass-filled retainer.

(3) Compare the results of the 15-weight-percent glass-filled retainer with those of the 30-weight-percent bronze-filled retainer of reference 7, and determine specifically whether equations of the type previously developed for the bronze-filled material could be applied to the glass-filled material.

The experimental study was conducted with 40-millimeter-bore ball bearings operating in cold hydrogen gas at 60° R (33 K) inlet temperature. The measured coolant-gas flow rates were from 0.008 to 0.045 pound force per second (0.0036 to 0.0204 kg/sec); shaft speeds were 20 000, 30 000, and 35 000 rpm; and thrust loads were 200, 300, and 400 pounds force (890, 1335, and 1780 N). The maximum Hertz stress varied from 215 000 to 290 000 psi (1.48×10^9 to 2.00×10^9 N/m²) at the inner-race contact and from 245 000 to 310 000 psi (1.7×10^9 to 2.14×10^9 N/m²) at the outer-race contact.

SYMBOLS

C_1, C_2	constants
K_1, K_2	constants in minimum flow equation
N	shaft speed, rpm
P	bearing thrust load, lbf (N)
T_1	temperature of coolant gas downstream from orifice flowmeter, $^{\circ}\text{R}$ (K)
T_2	temperature of coolant gas at inlet to bearing, $^{\circ}\text{R}$ (K)
T_3	temperature of coolant gas at outlet from bearing, $^{\circ}\text{R}$ (K)
T_4	temperature at outer diameter of bearing outer race, $^{\circ}\text{R}$ (K)
W	coolant-gas flow rate, lbm/sec (kg/sec)
W_{\min}	minimum flow rate of coolant gas, lbm/sec (kg/sec)

APPARATUS

Bearing Test Apparatus

The bearing test apparatus was the same as that first described in reference 5, and further used as noted in references 4 and 7. A schematic of the test-bearing mounting and housing is shown in figure 1. A variable-speed, direct-current motor drives the test-bearing shaft through a gear assembly. The bearing shaft speed was automatically controlled (within ± 0.1 percent) over the range of test conditions. The upper end of the test shaft was supported by an oil-lubricated ball bearing. The bearings were thrust loaded through a load collar (fig. 1) by applying deadweights to a lever arm attached to the housing.

Hydrogen Supply and Exhaust System

The test bearing was cooled by a direct stream of cryogenic hydrogen gas, as shown in figure 1. A schematic of the bearing coolant flow system is shown in figure 2. Liquid hydrogen supplied from a 500-gallon (2000-liter) Dewar was vaporized in a counterflow shell-and-tube heat exchanger by flowing hydrogen gas at ambient temperature through the shell. The cryogenic hydrogen-gas flow rate to the test bearing was measured by an orifice flowmeter located in the coolant-gas supply line downstream from the heat ex-

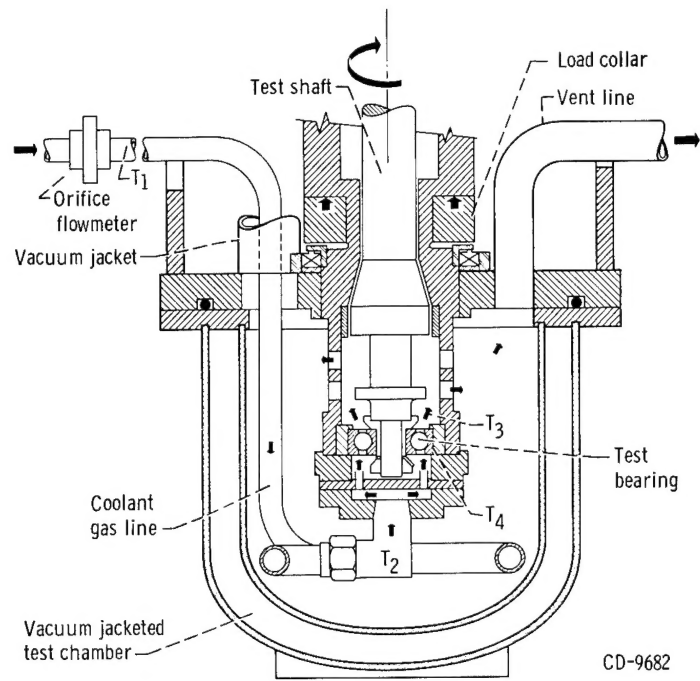


Figure 1. - Test-bearing mounting and coolant-gas supply.

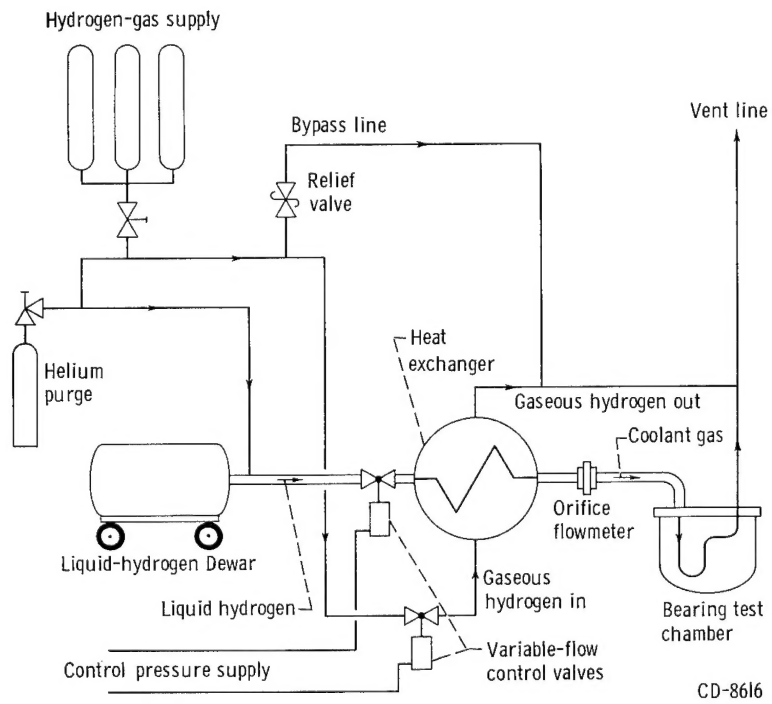


Figure 2. - Schematic of test-bearing coolant flow system.

changer. The range of the measured coolant-gas flow rate was from 0.008 to 0.045 pound mass per second (0.0036 to 0.0204 kg/sec). After flowing through the test bearing, the coolant gas was exhausted from the test chamber through the vent line. Both the liquid hydrogen flow from the Dewar and the warm gaseous hydrogen flow to the heat exchanger were regulated by remote-control variable-flow valves. The locations of these valves are shown in figure 2.

Temperature Measurement

Platinum resistance sensors were used to measure cryogenic temperatures at the following locations, as noted on figure 1:

- (1) Coolant-gas supply line downstream from the orifice flowmeter, T_1
- (2) Hydrogen-gas inlet to the test bearing, T_2
- (3) Hydrogen-gas exhaust downstream of the test bearing, T_3
- (4) Outer diameter of bearing outer-race, T_4

Temperatures could be read to an accuracy of $\pm 0.3 \text{ R}^\circ$ ($\pm 0.17 \text{ K}$) at 60° R (33 K) with the data recording system.

Test Bearings and Retainers

The test-bearing specifications are listed in table I. The bearings were 40-millimeter-bore (108 series), deep-groove type, manufactured to ABEC-5 tolerances.

TABLE I. - TEST-BEARING SPECIFICATIONS

Bearing type	Deep groove, separable at outer race
Bearing size	108 Series
Bearing grade	ABEC-5
Bore, mm	40
Ball diameter, in. (cm)	0.375 (0.95)
Number of balls	10
Ball and race material	AISI 440 C stainless steel
Inner-race curvature	0.54
Outer-race curvature	0.58
Radial clearance, in. (cm)	0.0025 (0.006)
Retainer material	15-percent glass-filled polytetrafluoroethylene (PTFE)
Retainer inner-land clearance, average, in. (cm)	0.026 or 0.038 (0.066 or 0.097)
Ball-pocket clearance, average, in. (cm)	0.015 or 0.020 (0.038 or 0.051)

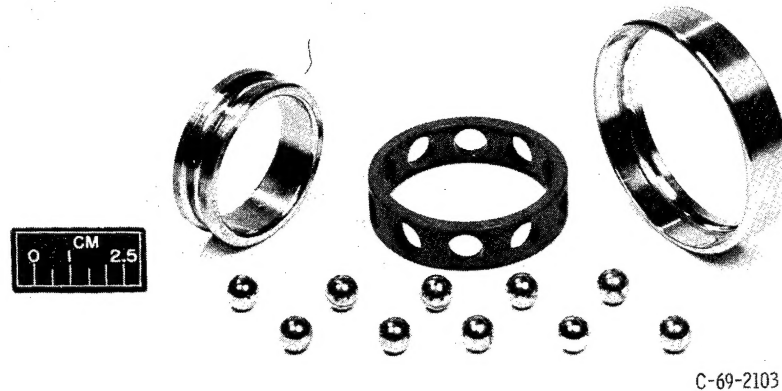


Figure 3. - Typical bearing set with one-piece machined retainer. Outer-race shoulder is removed for separation. Test-bearing specifications are listed in table I.

One shoulder on the outer race was relieved to make the bearings separable. The balls and races were AISI 440 C stainless steel.

The retainers were made of 15-weight-percent glass-fiber-filled PTFE material. They were of one-piece, machined construction and were inner-race located. The bearing components are shown in figure 3.

PROCEDURE

Pretest Procedure

To prepare the bearings for testing they were (1) degreased with trichloroethane, (2) inspected and measured for clearances, (3) checked with a surface-measuring instrument to obtain surface profiles of the inner-race grooves, (4) washed in trichloroethane, (5) stored in a vacuum-desiccator chamber for at least 6 hours, and (6) removed from the desiccator and weighed prior to testing.

Test Procedure

After the bearing was installed in the test housing, the test chamber and all hydrogen lines were purged for 15 minutes with helium gas. After the purge operation, cold hydrogen gas was pressure-fed to the bearing. The test shaft was rotated at 900 rpm during cooldown. The thrust load was applied early in the 10-minute cooldown period.

TABLE II. - SUMMARY OF MINIMUM FLOW RESULTS

Bearing	Thrust load		Speed, rpm	Constant, C ₁		Constant, C ₂		Correlation coefficient	Calculated minimum flow rate ^a		Calculated outer-race tempera- ture at minimum flow ^b	
	lbf	N		°R	K	(lb-°R)/sec	(kg-K)/sec		lb/sec	kg/sec	°R	K
4A	200	890	20×10 ³	58.4	32.4	0.0295	0.0074	0.713	0.0027	0.0012	69.3	38.5
	200	890	30	63.5	35.3	.1402	.0354	.994	.0059	.0027	87.3	48.5
	200	890	35	66.5	36.9	.4614	.1164	.997	.0107	.0049	109.6	60.6
7A	200	890	20×10 ³	60.9	33.8	0.0287	0.0072	0.954	0.0027	0.0012	71.5	39.6
	200	890	30	61.2	34.0	.2049	.0517	.942	.0072	.0033	89.6	49.7
17A	300	1335	20×10 ³	60.0	33.3	0.0762	0.0192	0.747	0.0044	0.0020	77.3	42.9
	300	1335	30	60.0	33.3	.1442	.0364	.989	.0060	.0027	84.0	46.6
	300	1335	35	44.0	24.4	.7285	.1837	.966	.0135	.0061	98.0	54.4
22A	300	1335	30×10 ³	58.2	32.3	0.1754	0.0442	0.991	0.0066	0.0030	84.8	47.1
23A	300	1335	20×10 ³	60.6	33.7	0.0369	0.0093	0.966	0.0030	0.0014	72.9	40.4
	300	1335	30	52.9	29.4	.4047	.1021	.993	.0101	.0046	93.0	51.6
	300	1335	35	47.4	26.3	.7813	.1971	.999	.0140	.0064	103.2	57.2
26A	400	1780	20×10 ³	60.4	33.6	0.0617	0.0156	0.978	0.0039	0.0017	76.2	42.3
	400	1780	30	55.2	30.7	.3460	.0873	.989	.0093	.0042	92.4	51.3
	400	1780	35	51.1	28.4	.5584	.1408	.850	.0118	.0054	98.4	54.6
27A	400	1780	20×10 ³	59.7	33.2	0.1178	0.0297	0.934	0.0054	0.0025	81.5	45.3
	400	1780	30	59.2	32.9	.3054	.0770	.978	.0087	.0040	94.3	52.3
	400	1780	35	39.2	21.8	.9415	.2375	.986	.0153	.0069	100.7	55.7
19A	200	890	30×10 ³	58.6	32.6	0.1538	0.0388	0.991	0.0062	0.0028	83.4	46.3
	300	1335	30	56.7	31.5	.2247	.0567	.883	.0075	.0034	86.7	48.2
	400	1780	30	57.2	31.8	.3460	.0873	.865	.0093	.0042	94.4	52.3
20A	200	890	30×10 ³	59.7	33.2	0.1592	0.0402	0.988	0.0063	0.0029	85.0	47.2
	300	1335	30	58.3	32.4	.2428	.0612	.987	.0078	.0035	89.4	49.7
	400	1780	30	49.7	27.6	.6847	.1727	.995	.0131	.0059	102.0	56.4
Composite data	200	890	20×10 ³	60.2	33.4	0.0150	0.0038	0.249	0.0019	0.0008	68.1	37.8
	200	890	30	60.0	33.3	.1896	.0478	.867	.0069	.0031	87.5	48.6
	200	890	35	66.5	36.9	.4614	.1164	.997	.0107	.0049	109.6	60.6
	300	1355	20	60.8	33.8	.0324	.0082	.776	.0028	.0013	72.4	40.3
	300	1355	30	56.2	31.2	.2681	.0676	.874	.0082	.0037	88.9	49.3
	300	1355	35	45.5	25.3	.7322	.1847	.961	.0135	.0061	99.7	55.3
	400	1780	20	59.8	33.2	.0998	.0252	.857	.005	.0023	79.8	44.3
	400	1780	30	54.5	30.3	.4236	.1068	.879	.0103	.0047	95.6	53.1
	400	1780	35	48.4	26.9	.6457	.1629	.870	.0127	.0058	99.2	55.0

^aCalculated from eq. (2).^bCalculated from eq. (1).

When the system approached temperature equilibrium, the shaft speed was increased to the test speed (in 5000-rpm increments every 5 min).

The flow rate of hydrogen gas was initially set at a high value, approximately 0.040 pound mass per second (0.018 kg/sec). This flow was maintained until temperature equilibrium was achieved in the system and the first data point taken, which required a total of 30 minutes. The coolant-gas flow rate was then lowered to obtain a new data point, while the shaft speed, thrust load, and inlet temperature T_2 were maintained constant. Temperature equilibrium for each data point was assumed when all the temperatures had remained constant for a minimum of 3 minutes, prior to recording the data. Total time at temperature equilibrium for each point was normally 5 minutes. After a very low hydrogen flow point had been obtained, a data point previously obtained at a high flow rate was repeated to check for bearing damage. Bearing outer-race temperature was normally repeatable to within 2 R^0 (1.1 K) at the high flow rates, unless bearing damage had occurred. This procedure was repeated for each speed and load condition.

The four cryogenic temperatures (T_1 , T_2 , T_3 , and T_4 , indicated on fig. 1) were recorded continuously throughout each bearing run. The 500-gallon (2000-liter) supply of liquid hydrogen provided enough coolant gas for a run time of approximately 120 minutes.

Temperature and flow rate data were thus obtained for each of seven bearings, run at a constant thrust load of either 200, 300, or 400 pounds force (890, 1335, or 1780 N) (table II) with shaft speeds of 20 000, 30 000 and 35 000 rpm. Data were also obtained for two bearings run at a constant shaft speed of 30 000 rpm and thrust loads of 200, 300, and 400 pounds force (890, 1335, and 1780 N) (table II).

The range of measured hydrogen coolant-gas flow rates was from 0.008 to 0.045 pound mass per second (0.0036 to 0.0204 kg/sec). The coolant-gas supply pressure, measured at the inlet to the test bearing, ranged from 14.6 to 15.6 psia (10.0×10^4 to $10.8 \times 10^4\text{ N/m}^2$).

Postrun Inspection of Bearings

After each run, the system was purged with helium gas. The test bearing was inspected for wear, washed in trichloroethane, and placed in a vacuum desiccator for approximately 6 hours.

At the end of a run series, each bearing was weighed to determine the weight loss or gain (in mg) of each component. Transverse surface profile traces were made on the inner-race grooves, in the region of the pretest surface profiles, to determine the extent of race wear and/or film buildup from transferred retainer material. The balls, races, and retainers were also examined qualitatively with a microscope at a magnification of 15 to help determine the extent of wear and surface damage.

RESULTS AND DISCUSSION

Experimental Test Results

Effect of coolant flow rate on bearing outer-race temperature. - The investigation was conducted to determine the relation between bearing outer-race temperature T_4 and coolant flow rate W . A total of nine ball bearings with 15-weight-percent glass-fiber-filled PTFE retainers were run at constant speed and load conditions while flowing hydrogen gas at 60°R (33 K) through the bearings (table II). The experimental results for seven bearings run at a constant thrust load of either 200, 300, or 400 pounds force (890, 1335, or 1780 N) with shaft speeds of 20 000, 30 000, and 35 000 rpm are shown in figures 4(a) to (g). The results for two bearings run at a constant shaft speed of 30 000 rpm and thrust loads of 200, 300, and 400 pounds force (890, 1335, and 1780 N) are shown in figure 5. As would be expected, the outer-race temperature T_4 increased as the coolant flow rate decreased. Also, T_4 was generally higher at higher shaft speeds and/or higher thrust loads.

In reference 7, a simplified heat-transfer analysis showed that the relation between bearing outer-race temperature and coolant flow rate could be expressed as

$$T_4 = C_1 + \frac{C_2}{W} \quad (1)$$

Therefore, the curves of figures 4 and 5 were determined by a least-squares fit of equation (1) with the experimental data. The curves were extrapolated to flow rates beyond the experimental data, which are shown as the dashed portions of the curves.

Constants C_1 and C_2 and the correlation coefficients for the curves of figures 4 and 5 are given in table II. The correlation coefficient is defined as the ratio of the explained variance to the total variance of the data (see ref. 9). A curve fit is generally considered to be adequate if the correlation coefficient is greater than 0.8. An indication that equation (1) provided a good representation of the data was that most of the correlation coefficients were above 0.90.

In figures 4 and 5, results from individual bearings were used to verify the relation between outer-race temperature and coolant flow rate so that the effect of data scatter would be minimized. To observe scatter in the data due to differences among the individual bearings, the data from all bearings tested at the same condition of speed and load were plotted on one graph, an example of which is shown in figure 6. Curves for each speed and load condition tested were obtained by a least-squares fit of the composite data with equation (1). The correlation coefficients and constants for these curves are

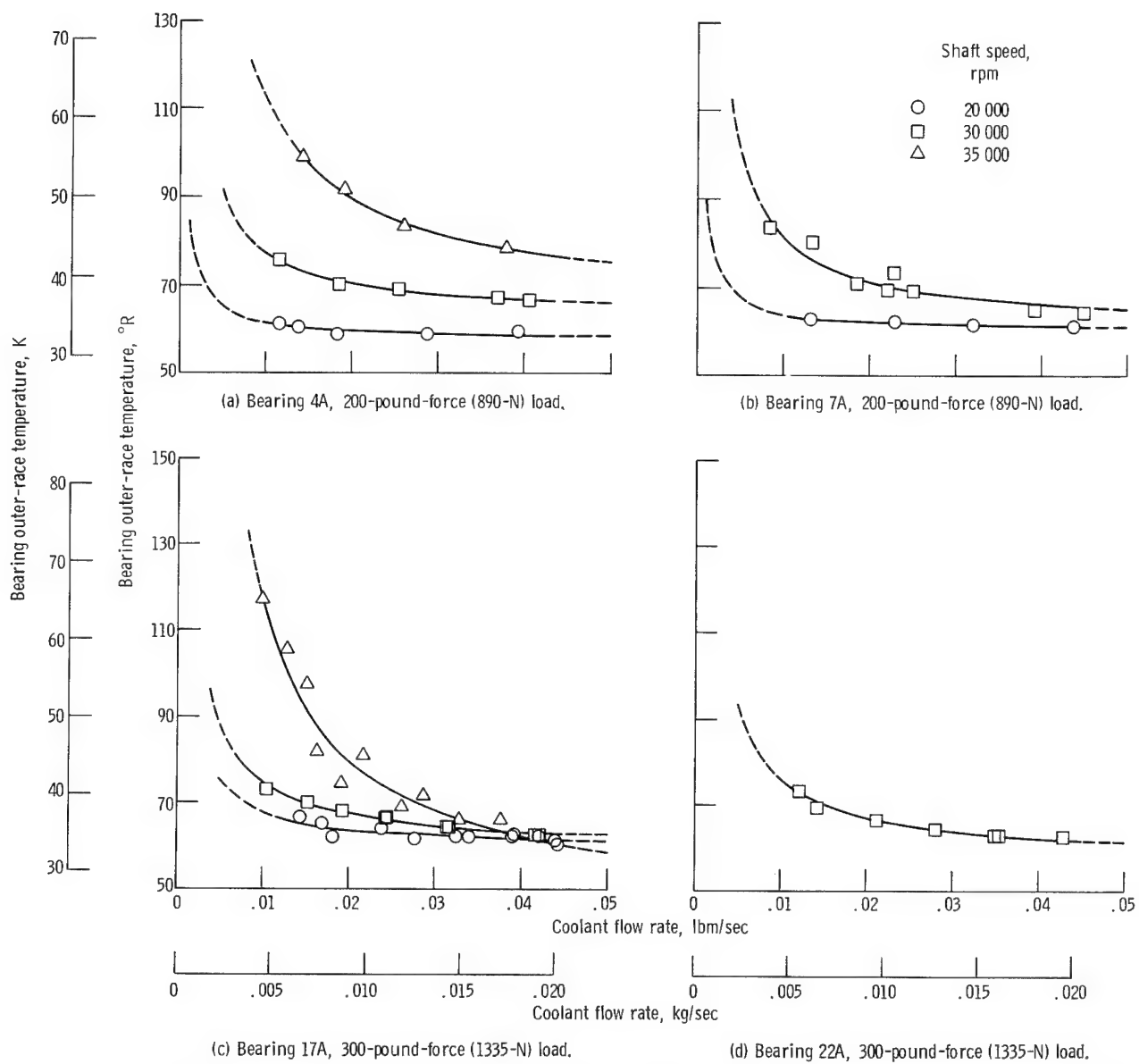
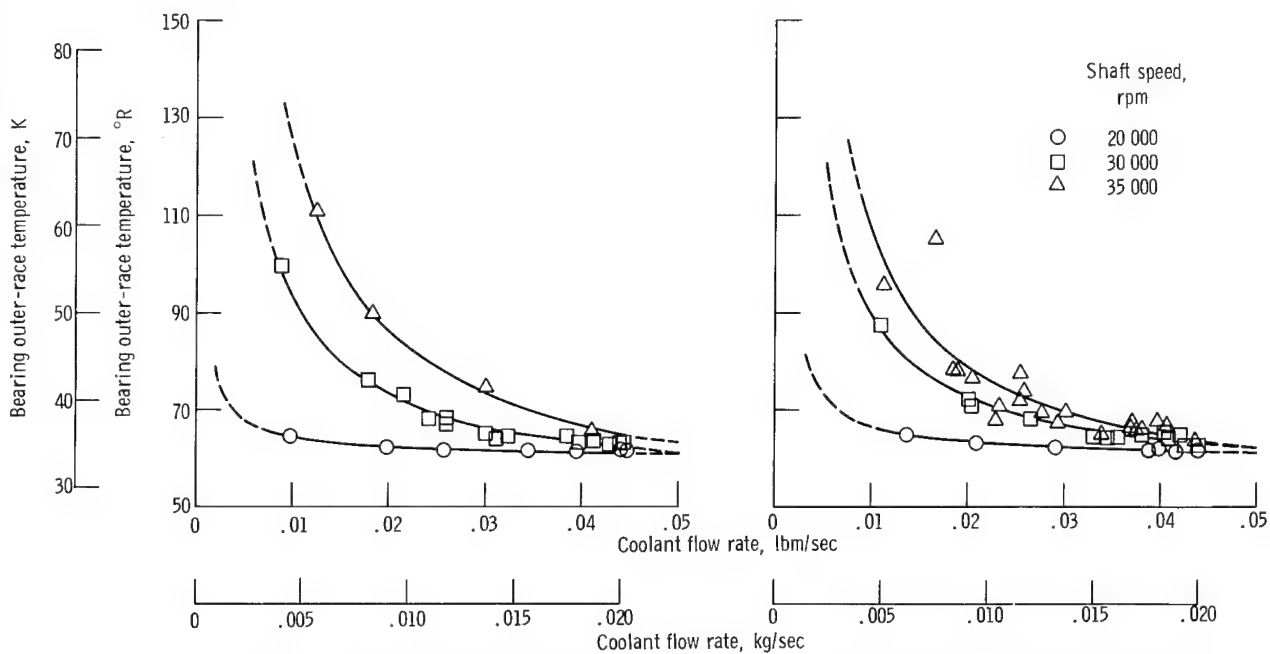
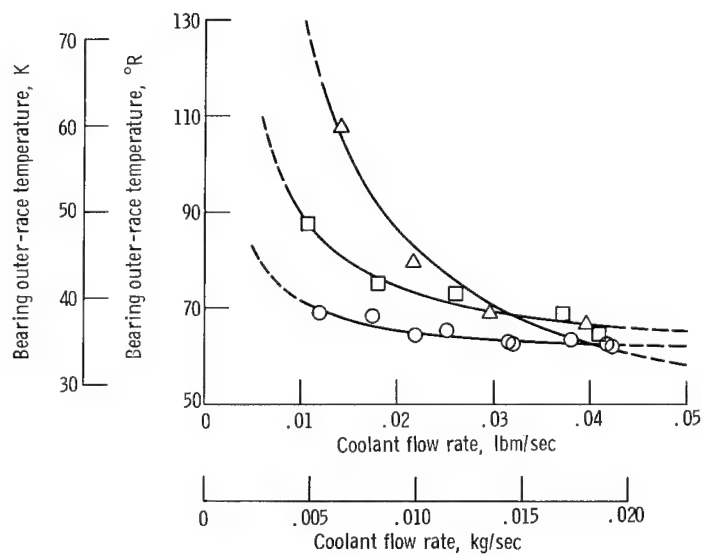


Figure 4. - Bearing outer-race temperature as function of hydrogen-gas coolant flow rate for bearings tested with constant thrust load. (The dashed parts of the curves denote extrapolated data.)



(e) Bearing 23A, 300-pound-force (1335-N) load.

(f) Bearing 26A, 400-pound-force (1780-N) load.



(g) Bearing 27A, 400-pound-force (1780-N) load.

Figure 4. - Concluded.

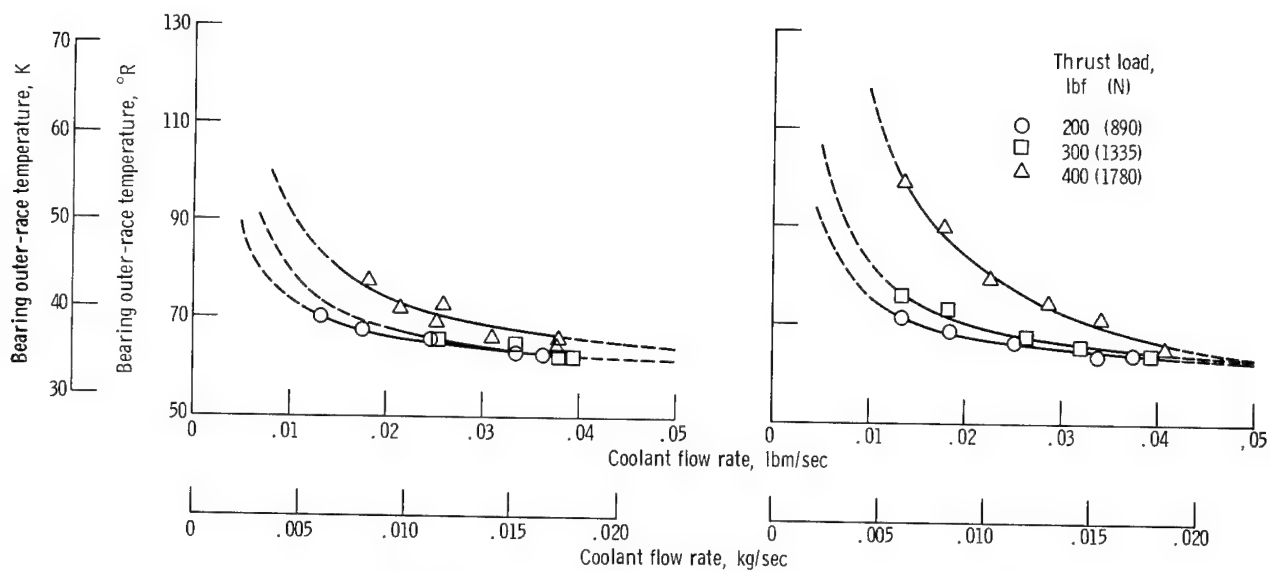


Figure 5. - Bearing outer-race temperature as function of hydrogen-gas coolant flow rate for bearings tested at constant shaft speed of 30 000 rpm with three thrust loads. (The dashed parts of the curves denote extrapolated data.)

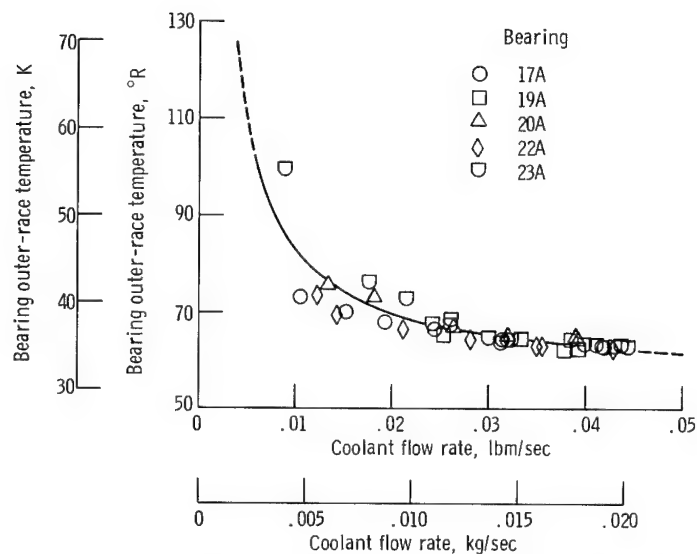


Figure 6. - Bearing outer-race temperature as function of hydrogen-gas coolant flow rate for all bearings tested at 30 000-rpm shaft speed and 300 pounds force (1335 N) thrust load. (The dashed parts of the curves denote extrapolated data.)

given at the bottom of table II. The correlation coefficients were understandably lower for the composite data from several bearings than for the data from a single bearing. Most of the coefficients were still above 0.85, however.

Bearing temperature instability. - During testing, sporadic excursions of the bearing temperature occurred. A trace of the outer-race temperature during a typical instability is shown in figure 7. Fluctuations in shaft speed and drive-motor current occurred simultaneously with those of the bearing outer-race temperature. After the excursions, the bearing temperature returned to its previous stabilized value. The retainer land clearance was increased on bearing 23A, but the temperature instability still occurred. These excursions did not occur on every run, but did happen eventually with every bearing that had a ball-pocket clearance of 0.015 inch (0.038 cm), except with bearing 22A. However, this bearing was tested only once for a total of 104 minutes. Most of the other bearings did not experience temperature excursions on the first run, so bearing 22A was not unusual.

The ball-pocket clearance was therefore increased to 0.020 inch (0.05 cm) for further tests. Bearing 23A (already run for 376 min with 0.015 in. (0.038 cm) clearance) was tested for an additional 448 minutes with the ball-pocket clearance of 0.020 inch (0.05 cm) with no further temperature excursions. Bearing 26A ran a total of 946 minutes with only two small excursions at 412 minutes of run time. Bearing 27A ran a total of 524 minutes; some temperature excursions occurred during one startup, after about 316 minutes of run time. Increasing the ball-pocket clearance tended to eliminate the

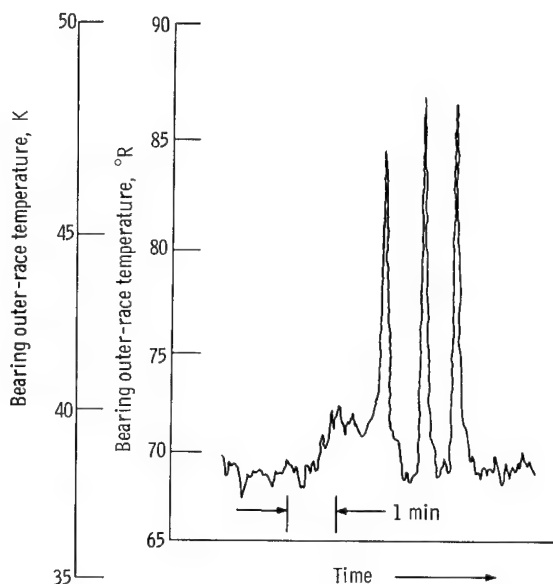


Figure 7. - Record of outer-race temperature sensor during a typical temperature instability. Bearing load, 300 pounds force (1335 N); shaft speed, 30 000 rpm; bearing 23A.

TABLE III. - RETAINER CLEARANCE

IDENTIFICATION

Bearing	Retainer			
	Inner-land clearance		Ball-pocket clearance	
	in.	cm	in.	cm
4A	0.026	0.066	0.015	0.038
7A	↓	↓	↓	↓
17A	↓	↓	↓	↓
19A	↓	↓	↓	↓
20A	↓	↓	↓	↓
22A	↓	↓	↓	↓
23A	.038	.097	0.015, 0.020	0.038, 0.051
26A	.026	.066	.020	.051
27A	.038	.097	.020	.051

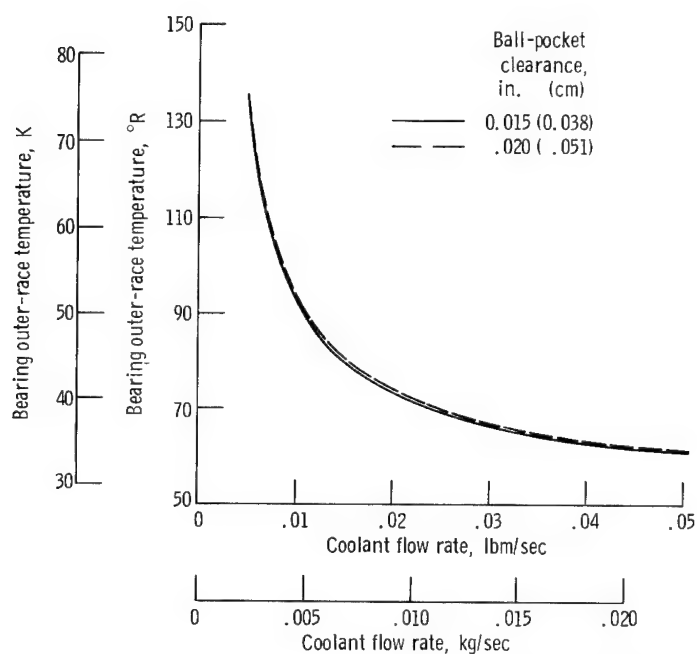


Figure 8. - Effect of enlarging cage ball-pocket diameter on relation between outer-race temperature and coolant flow rate. Shaft speed, 30 000 rpm; thrust load, 300 pounds force (1335 N); bearing 23A.

sporadic temperature instability. The retainer ball-pocket and inner-land clearances for each bearing are shown in table III.

Since the outer-race temperature returned to a normal value after these excursions, increasing the ball-pocket clearance should have no effect on the temperature - coolant flow relation. This fact is shown in figure 8, where the curves represent the data taken for bearing 23A with ball-pocket clearances of 0.015 and 0.020 inch (0.038 and 0.051 cm). Enlarging the ball pocket had virtually no effect on the response of the bearing outer-race temperature to changes in coolant flow rate.

Minimum Coolant Flow Rate

Definition and calculation of minimum flow rate. - The minimum coolant flow rate was defined in reference 7 as the lowest flow rate that should be used to operate a bearing at a given speed and load condition. For that investigation, the criterion selected was the sensitivity of the bearing outer-race temperature to changes in coolant flow rate, or dT_4/dW . The minimum flow rate, then became that flow at which the sensitivity dT_4/dW became critical. The minimum flow rates for reference 7 were thus calculated from the following equation, derived in the reference:

$$W_{\min} = \sqrt{\frac{C_2}{\left| \frac{dT_4}{dW} \right|_{\max}}} \quad (2)$$

In reference 7, a value of 4000 R^0 per pound mass per second (4890 K/kg/sec) was selected as the critical sensitivity.

In the present work, it was decided to define and calculate the minimum flow rate by the method used in reference 7. Therefore, the minimum flow rates were calculated from equation (2). Values of C_2 were determined for each speed and load condition from the least-squares analysis, as noted previously, and 4000 R^0 per pound mass per second (4890 K/kg/sec) was used for $\left| dT_4/dW \right|_{\max}$. The results are presented in table II. The outer-race temperatures corresponding to the minimum flow rates were calculated from equation (1) and are also given in table II.

Effect of shaft speed and thrust load on minimum flow rate. - The minimum flow rate, calculated as described in the preceding section, is shown in figure 9 as a function of shaft speed for a constant thrust load. The minimum flow rate increases quite rapidly with shaft speed.

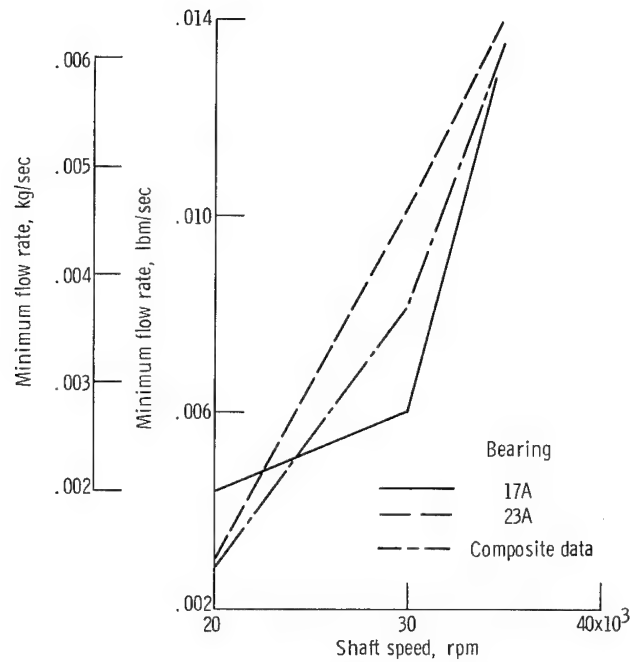


Figure 9. - Effect of shaft speed on required minimum hydrogen-gas coolant flow rate. Load, 300 pounds force (1335 N).

In figure 10, the minimum flow rate is shown as a function of thrust load for a constant shaft speed. The effect of load (fig. 10) is seen to be smaller than the effect of speed (fig. 9).

In order for the minimum flow rate equation (eq. (2)) to be applicable for any bearing operating conditions over the range investigated, the constant C_2 must be expressed in terms of shaft speed N and thrust load P . This relation, determined empirically in the appendix, shows that

$$C_2 \propto PN^5$$

so that equation (2) becomes

$$W_{\min} = K_2 \sqrt{PN^5} \quad (3)$$

The final equation then, also derived in the appendix, for minimum coolant flow rate, for 40-millimeter-bore ball bearings with 15-weight-percent glass-filled PTFE retainers, cooled by 60° R (33 K) hydrogen gas over the range of operating conditions investigated, can be written as

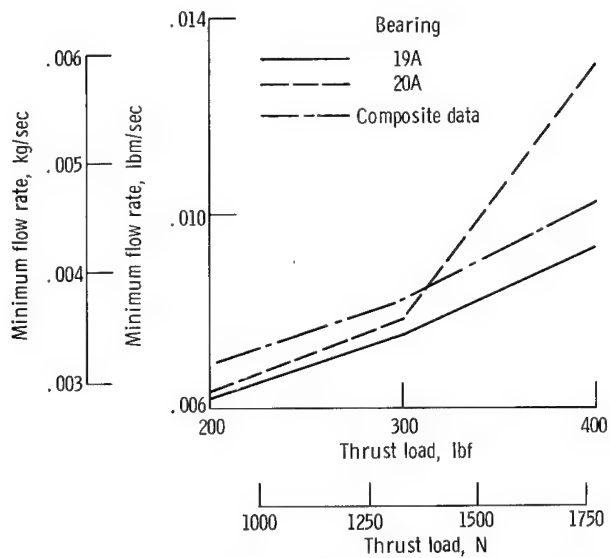


Figure 10. - Effect of thrust load on minimum hydrogen-gas coolant flow rate. Shaft speed, 30 000 rpm.

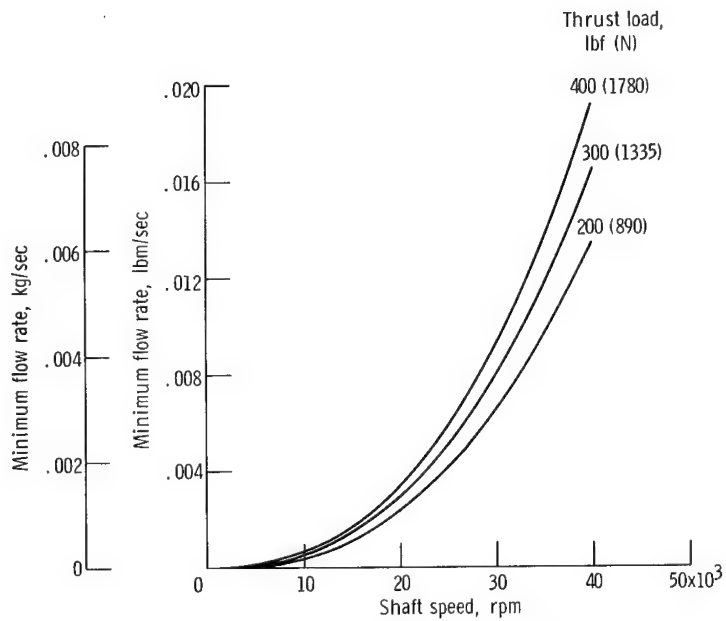


Figure 11. - Calculated minimum hydrogen-gas coolant flow rate as function of shaft speed for three thrust loads (eq. (4)).

$$\left. \begin{array}{l} \text{or} \\ W_{\min} = 3.0 \times 10^{-15} \sqrt{PN^5} \quad \text{lbm/sec} \\ W_{\min} = 0.64 \times 10^{-15} \sqrt{PN^5} \quad \text{kg/sec} \end{array} \right\} \quad (4)$$

Minimum flow rate as a function of shaft speed for three loads, as calculated from equation (4), is shown in figure 11.

Postrun Examination

At the end of the run series, as stated previously, the components of each bearing were examined with a microscope at a magnification of 15. Film transfer or surface damage on the balls and races and the degree of retainer wear were observed. The total run times, retainer wear, sliding distance of the retainer at the inner race, and postrun conditions of the bearings are given in table IV.

Inner race. - Profile traces were made of the inner-race grooves to indicate the postrun condition. A typical profile trace of the inner race in the ball-track area is shown in figure 12. As noted in table IV, most of the bearings were still in good condition, although several showed slight wear. The film transfer characteristics were similar to those reported in reference 4, as indicated by profile traces for the same retainer material.

Retainer wear. - The retainer wear is given in table IV as the percent loss for the total run times shown. The wear was very light in all bearings, although bearing 4A showed some heavy wear in two ball pockets. This damage occurred during an attempt to obtain data at 40 000 rpm. When the bearings are compared on the basis of retainer wear rate (percent weight loss divided by the total sliding distance, table IV), little difference exists among the bearings, with the exception of bearing 4A. The wear rates are quite low.

The bearings in a rocket engine turbopump are required to operate for only about 4 or 5 hours total running time (ref. 7). The bearings in references 4 and 7 were run for as long as 10 hours. Bearing 17A, however, was run for a total of over 21 hours, and it is interesting to look at the wear history of this bearing, as shown in figure 13. The wear of the retainer seems to increase constantly with the sliding distance. The sum of the weight losses of the ball set and retainer is more than the total bearing weight loss for a sliding distance of about 8.3 million feet (2.5×10^6 m). This shows that some of the retainer material has transferred to the races. Between 8.3 and 8.9 million feet (2.5×10^6 and 2.7×10^6 m), the bearing had started to wear out, as evidenced by the sharp

TABLE IV. - BEARING POSTRUN CONDITION

Bearing	Load		Total run time, min	Retainer				Postrun condition		
	lbf	N		Total wear, wt. %	Sliding distance at inner race		Wear rate, wt. % per 10 ⁶ ft (0.3×10 ⁶ m)	Retainer	Inner race	Balls
					ft	m				
4A	200	890	466	1. 0184	3. 70×10 ⁶	1. 13×10 ⁶	0. 28	Heavy wear, two ball pockets	Very wide film area, wear evident	Track visible on one ball only
7A	200	890	592	0. 1177	4. 44×10 ⁶	1. 35×10 ⁶	0. 026	Low wear	Wide film area, wear evident	Multiple tracks, no visible wear
17A	300	1335	1270	0. 1539	8. 86×10 ⁶	2. 70×10 ⁶	0. 017	Radial scratches in ball pockets, low wear	Wide film area, wear evident	Visible wear track
22A	300	1335	104	0. 0236	0. 84×10 ⁶	0. 26×10 ⁶	0. 028	Low wear	Narrow film area	Clean
23A	300	1335	824	0. 1554	5. 91×10 ⁶	1. 80×10 ⁶	0. 026	Low wear	Wide film area, wear evident	Multiple tracks, visible wear
26A	400	1780	946	^a 0. 0736	6. 90×10 ⁶	2. 10×10 ⁶	^a 0. 021	Low wear	Wide film area	Ultrasonically cleaned, track barely visible
27A	400	1780	524	^b 0. 2405	3. 38×10 ⁶	1. 03×10 ⁶	^b 0. 091	Low wear	Very wide film area, wear evident	Ultrasonically cleaned, visible wear track
19A	200 300 400	890 1335 1780	} 412	0. 1241	3. 14×10 ⁶	0. 96×10 ⁶	0. 039	Low wear	Wide film area	Multiple tracks, no visible wear
20A	200 300 400	890 1335 1780	} 421	0. 0505	3. 18×10 ⁶	0. 97×10 ⁶	0. 016	Low wear	Wide film area	Multiple tracks, visible wear

^aAfter 445 min.^bAfter 405 min.

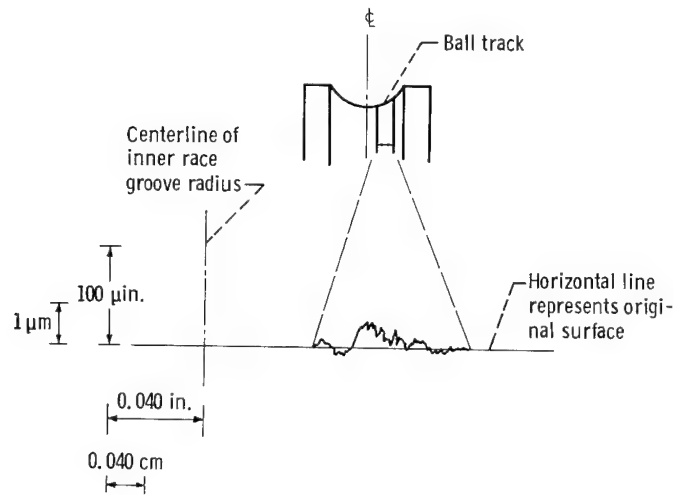


Figure 12. - Profile trace of inner race in ball-track area after completion of test runs, compared to original surface. Bearing 26A; shaft revolutions, 23.9×10^6 ; thrust load, 400 pounds force (1780 N).

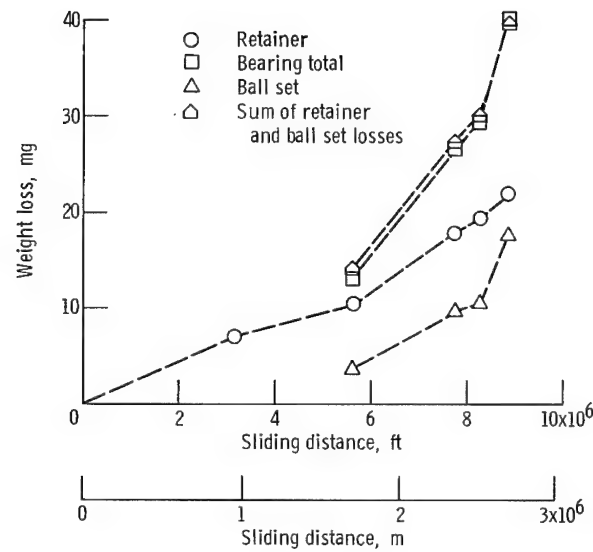


Figure 13. - Wear history of bearing 17A. Thrust load, 300 pounds force (1335 N); shaft speeds, 20 000, 30 000, and 35 000 rpm; retainer sliding on inner race.

increase in ball set wear at 8.9 million feet (2.7×10^6 m). Race wear had increased at this point also, since the sum of the ball set and retainer losses was less than the total bearing weight loss.

Comparison of Results from Two Filled PTFE Retainer Materials

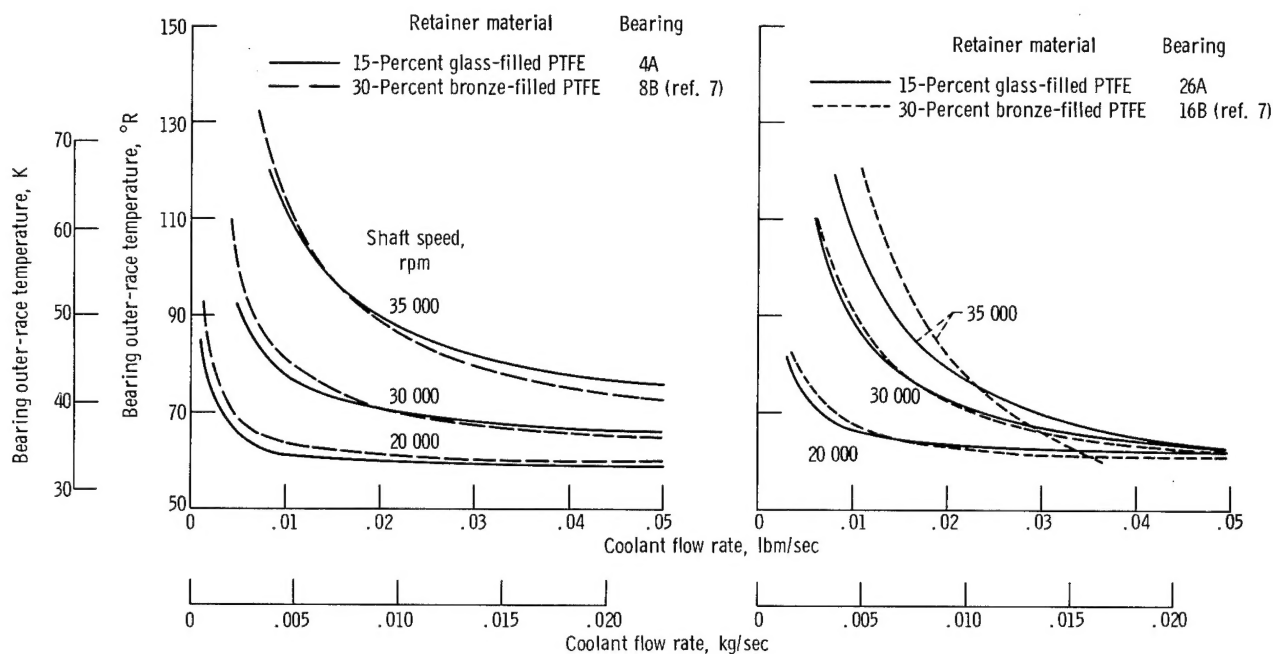
It is informative to compare the results of the present work using a 15-weight-percent glass-filled PTFE retainer material with the results from reference 7, where the retainer material was 30-weight-percent bronze-filled PTFE. The relation between the outer-race temperature and the coolant flow rate generally compared very well for the two materials. A typical comparison is shown in figures 14(a) and (b) for three shaft speeds at constant loads of 200 and 400 pounds force (890 and 1780 N).

As would be expected, since the outer-race temperature - coolant flow rate relations agreed so well, the minimum flow equations actually compare very well also, over the range of test conditions investigated. The best fit of the data showed that the constant C_2 was a function of shaft speed N to the fifth power with the glass-filled PTFE (see appendix) and to the fourth power for the bronze-filled PTFE (ref. 7). However, if the data for the glass-filled PTFE retainer bearings are fit as a function of speed N to the fourth power, the equation for minimum flow rate becomes

$$\left. \begin{array}{l} \text{or} \\ W_{\min} = 0.5 \times 10^{-12} N^2 \sqrt{P} \quad \text{lbm/sec} \\ W_{\min} = 0.11 \times 10^{-12} N^2 \sqrt{P} \quad \text{kg/sec} \end{array} \right\} \quad (5)$$

Compare this with the following equation for the bronze-filled PTFE retainer from reference 7:

$$\left. \begin{array}{l} \text{or} \\ W_{\min} = 0.575 \times 10^{-12} N^2 \sqrt{P} \quad \text{lbm/sec} \\ W_{\min} = 0.118 \times 10^{-12} N^2 \sqrt{P} \quad \text{kg/sec} \end{array} \right\} \quad (6)$$



(a) Thrust load, 200 pounds force (890 N). (b) Thrust load, 400 pounds force (1780 N).
Figure 14. - Comparison of test results using two retainer materials - 15-percent glass-filled PTFE (present work) and 30-percent bronze-filled PTFE (ref. 7).

It is suggested by the authors that equation (6) can be used to calculate minimum flow rate of hydrogen gas at 60°R (33 K) for bearings using either of the two materials, operating under the conditions investigated.

Bearing wear, however, was somewhat different for the two retainer materials. The retainer wear rate was only about 0.03 percent per million sliding feet ($0.3 \times 10^6 \text{ m}$) for the glass-filled PTFE, but was from 0.5 to 1.0 percent for the bronze-filled PTFE. There was no evidence of wear on the inner race with the bronze-filled PTFE; however, inner-race wear was evident with several bearings using the glass-filled PTFE. A possible reason for the difference is that the glass is more abrasive than the bronze. Both retainer materials did provide a transfer film for satisfactory bearing operation for extended running times, as shown in table IV of this report and in table II of reference 7.

SUMMARY OF RESULTS

The cooling requirements of 40-millimeter-bore ball bearings operating in hydrogen gas at 60°R (33 K) were investigated at thrust loads of 200, 300, and 400 pounds force (890, 1335 and 1780 N) and shaft speeds of 20 000, 30 000, and 35 000 rpm. The bearing retainer material was a 15-weight-percent glass-fiber-filled PTFE. The hydrogen coolant-gas flow rates through the bearings ranged from 0.008 to 0.045 pound mass per

second (0.0036 to 0.0204 kg/sec). The minimum required flow rate of hydrogen-gas coolant through the bearing for a given speed and load condition was determined by the response of the bearing outer-race temperature to changes in coolant-gas flow rate. The study produced the following results:

1. From the experimental results, an empirical equation was developed to determine the minimum flow rates of coolant gas required for thermally stable operation of the bearings with the glass-filled PTFE retainers over the range of operating conditions investigated. The equation is of the form previously developed for bearings with a bronze-filled PTFE retainer material. The outer-race temperature - coolant flow relation and the minimum flow equations were similar for both materials.

2. The retainer wear rate was much less for the glass-filled material than for the bronze-filled material; however, race wear was evident for some bearings with the glass-filled retainer.

3. Minimum flow requirements for the bearings were more sensitive to increases in speed than to increases in thrust load.

4. Evidence of the formation and buildup of transfer films on the inner race from the retainer material was provided by profile traces of the bearing inner-race ball track taken before starting and after completion of test runs.

5. Increasing the ball-pocket clearance tended to eliminate a sporadic temperature instability encountered during some of the tests. Enlarging the ball pocket had virtually no effect on the outer-race temperature - coolant flow rate data.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, October 15, 1969,
129-03.

APPENDIX - EMPIRICAL DETERMINATION OF MINIMUM FLOW EQUATION

The minimum flow rate was defined in the section RESULTS AND DISCUSSION and in reference 7 as the lowest flow rate that should be used to operate a bearing at a given speed and load condition. It was shown in reference 7 that this minimum flow rate could be expressed as

$$W_{\min} = \sqrt{\frac{C_2}{\left| \frac{dT_4}{dW} \right|_{\max}}} \quad (2)$$

This equation was used in the present report to calculate the minimum flow rates, using a value of 4000 R⁰ per pound per second (4890 K/kg/sec) for $\left| dT_4/dW \right|_{\max}$. Values of C_2 were obtained from the least-squares analysis and are given in table II. However, to obtain a more generalized correlation for minimum flow rate, for bearings using a 15-weight-percent glass-filled PTFE retainer material, the dependence of C_2 on shaft speed N and thrust load P must be established. This dependence was determined empirically by plotting C_2 against speed at constant load and C_2 against load at constant speed on log-log coordinate paper. The slope of each curve is the power of the speed and load dependence, respectively. These plots showed that $C_2 \propto PN^5$ so that equation (2) becomes

$$W_{\min} = \sqrt{\frac{K_1 PN^5}{\left| \frac{dT_4}{dW} \right|_{\max}}} \quad (7)$$

The value of K_1 was determined by plotting C_2 as a function of PN^5 on log-log paper. When a line with a slope of 1 was positioned through the data, K_1 was determined to be 3.546×10^{-26} R⁰-min⁵/sec (4.33×10^{-26} K-min⁵/sec). Therefore, using this value of K_1 and a $\left| dT_4/dW \right|_{\max}$ of 4000 R⁰ per pound mass per second (4890 K/kg/sec) results in the following equation for minimum flow rate:

$$\left. \begin{aligned} W_{\min} &= 3.0 \times 10^{-15} \sqrt{PN^5} \quad \text{lbm/sec} \\ W_{\min} &= 0.64 \times 10^{-15} \sqrt{PN^5} \quad \text{kg/sec} \end{aligned} \right\} \quad (4)$$

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